INVESTIGATION AND EFFICIENCY MAXIMIZATION OF THE OPERATION AND DESIGN OF A SMALL SCALE EXPERIMENTAL TRIGENERATION SYSTEM POWERED BY A SUPERCRITICAL ORC

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ABSTRACT

In this work, the detailed investigation and the optimization of the operational parameters of an experimental, small scale trigeneration system encompassing a supercritical Organic Rankine Cycle (ORC) and a heat pump are presented. Both the ORC and the heat pump jointly operate with the same working fluid (R227ea). The heat input to the ORC is provided by a 85 kW_{th} biomass boiler. The electricity produced by the ORC, which has a nominal power output of 5 kW_e, is used to power the heat pump, capable of covering a cooling load of 4 kW_{th}, while any surplus electricity is exported to the grid. Meanwhile, the heat generated during the condensation of the working fluid (around 70 kW_{th}) is utilized to produce hot water. The system has therefore the potential to produce combined cooling, heating and electricity, depending on the load requirements, by utilizing a renewable energy source with zero net CO_2 emissions.
The investigation carried out includes the selection process of the working fluid of the system through the comparison of its performance with that of other typical working fluids and by taking into account environmental and safety factors. Furthermore the study presents the optimization procedure for selecting the working temperatures and pressures in order to maximize the cycle’s efficiency, given the technological limits of the elements of this system (heat exchangers, scroll expanders etc.). In addition, a supercritical plate heat exchanger model, used for the design of the heat exchanger of the unit, is presented.

1. INTRODUCTION

In the recent years, a significant amount of interest has been focused on multigeneration systems aiming to convert a primary energy source (fossil, solar, waste heat) into combined electricity, cooling and heating. This is in part driven by the fact that traditional electricity generation systems have a restricted efficiency (around 30-40 % [1], [2]), so a great deal of the original heat is rejected to the environment unexploited in the form of waste heat. Due to the policies followed worldwide in order to increase overall system efficiencies and restrain the emissions of greenhouse gas (GHGs), a lot of research focuses on the design of cost competitive co-generation and combined cooling, heating and power systems is carried out. Furthermore, small scale cogeneration and trigeneration systems that use renewable energy sources have gathered significant attention, since they can potentially contribute to a further reduction of emissions, while also ensuring sustainability and fuel independence.
The Organic Rankine Cycle (ORC) has lately gathered substantial interest as a promising technology in the field of power generation from low temperature heat sources, such as solar and geothermal energy and industrial waste heat. The traditional water-steam Rankine cycle, implemented in the conventional high-temperature thermal power plants, is in many cases not economic or technically feasible. This is because its implementation is not favorable for low grade applications and small-scale power outputs [3], such as those encountered in solar thermal plants. The Organic Rankine

Cycle (ORC), on the other hand, poses certain advantages compared to the conventional cycle. Some of these include the potential of low temperature heat recovery due to the lower boiling point of the working fluids used, the overall smaller component size as well as the capability of expander operation under smaller temperatures [4]. Moreover, due to the "dry" organic fluids having a positive dT/dS saturation vapor line, it is not in principle necessary to superheat them [5]. Meanwhile, the supercritical ORC (SORC), in which the working fluid is pressurized to supercritical pressures before its entrance to the heater has been shown to exhibit several advantages, such as improved thermal and also exergetic efficiency [6-8]. The most important characteristic of supercritical ORCs is the fact that, because of the supercritical heating pressures, the working fluid does not gradually evaporate into the gaseous phase. Instead, it changes from the liquid to the supercritical state when its temperature increases above its critical value. Despite, the fact that, compared to a subcritical, the supercritical ORC results in higher operating pressures in the cycle, it is worthy evaluating its competiveness for small scale systems in a practical way.

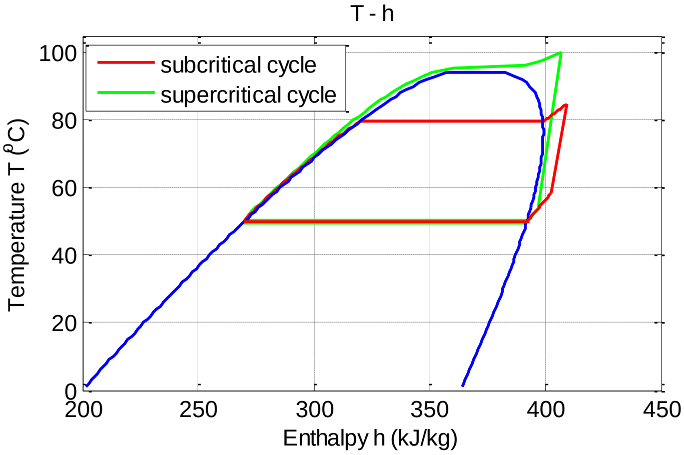

Figure 1: Sub- and supercritical ORC. Example of R1234yf

In this study an experimental trigeneration unit currently under construction in the Laboratory of Steam Boilers and Thermal Plants (LSBTP) in the National Technical University of Athens (NTUA), Greece, is introduced. The system consists of a supercritical Organic Rankine Cycle (SORC) interconnected with a Vapor Compression Cycle (VCC). Heat is provided to the system by a biomass boiler. The two systems are capable of combined electricity, cooling and heating generation, as can be seen in the simplified process diagram of Figure 2. In the present work, the preliminary design and decision process regarding the working fluid selection, the adjustment of some key thermodynamic operational parameters and the selection of equipment is presented and justified.

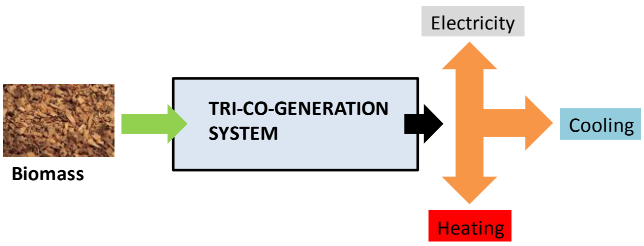

Figure 2: Simple process diagram of the ORC-VCC system under construction

2. SYSTEM DESCRIPTION

2.1 SORC and VCC modules

The system under design and construction consists of three main subsystems, the SORC module, the VCC module and the biomass boiler-heating oil module. The SORC is interconnected to the VCC and both cycles use the same working fluid. The SORC includes a pump and a series of heat exchangers, where energy is transferred from the heat sources to the working fluid. After reaching its maximum temperature the working fluid flows through the expander, producing useful work, which is used to cover the power needs of the system, while any surplus is exported to the grid

The supercritical fluid exiting the expander of the SORC is mixed with the stream exiting the compressor of the VCC and the mixture both enter the condenser of the system. In this way, the condensation step of both cycles takes place in a single condenser under a common pressure, providing the heat output of the system which is used for producing hot water. Apart from the compressor, the VCC is comprised of an expansion valve and an evaporator, where it absorbs heat, thus generating the cooling output of the system.
The power required for the compression of the cooling cycle is provided by the generator of the SORC, while any surplus power can be exported to the electricity grid.
During the winter, when there is no demand for cooling, the VCC can be disconnected from the rest of the system, giving an increased potential for electricity generation. In this manner, the operation mode can alternate between trigeneration and cogeneration. On trigeneration mode, it is necessary that the power produced by the ORC is adequate for the operation of the compressor in order for the system to be independent from external power sources. It must be mentioned, that this system is to operate at steady state conditions, besides the inevitable fluctuations of the biomass boiler.

2.2 Biomass boiler-heating oil circuit

The heat input to the SORC originates from the combustion of biomass in a boiler. The combustion heat is provided to a heating oil which then flows through an intermediate heat transfer loop (HTL) in order to deliver its energy content to the working fluid in the heater of the SORC. The heating oil then returns to the biomass boiler to increase its temperature.

3. MATHEMATICAL MODELING AND SIMULATION

3.1 Mathematical formulation

The first step of the design of the experimental unit is its thermodynamic modeling, which will subsequently allow to select the optimal values of some of its key operational parameters. The simulation of the system is carried out by numerically solving a set of mathematical equations that describe the operation of its components. For the simulation of the system, a steady state operation is assumed, while heat and pressure losses along the equipment are neglected. The equilibrium thermodynamic properties of the working fluid in each state are calculated with the CoolProp database [9] incorporated in the Matlab Software [10]. The most important system performance evaluation indexes used for the design of the facility are the thermal efficiency of the SCORC $\eta_{th,ORC}$, the electrical efficiency of the ORC $\eta_{el,ORC}$, the Coefficient of Performance (COP) of the VCC, the heat production efficiency of the system $\eta_{heat,sys}$ as well as its overall cogeneration efficiency $\eta_{CHP,sys}$. These are described by the following equations:

$$\eta_{th,ORC} = \frac{P_{turb} - P_{pump}}{Q_{ORC,in}} \tag{1}$$

$$\eta_{el,ORC} = \frac{P_{el,net}}{Q_{ORC,in}} \tag{2}$$

$$COP = \frac{Q_{cool}}{P_{el,comp}} \tag{3}$$

$$\eta_{heat,sys} = \frac{Q_{cond}}{Q_{ORC,in}} \tag{4}$$

$$\eta_{\mathrm{CHP},sys} = \frac{Q_{cond} + P_{el,net}}{Q_{ORC,in}} \tag{5}$$

In the above equations, P_{turb} is the work derived in the turbines, P_{pump} is the work consumed in the pump, $P_{el,net}$ is the net electric output of the SORC expander, Q_{cool} is the cooling duty of the VCC, $P_{el,comp}$ is the electricity consumed by the VCC compressor, Q_{heat} is the heat generated in the condenser of the system and $Q_{ORC,in}$ is the heat provided to the working fluid in the heater of the SORC.

3.2 Assumptions

Further assumptions made regarding the operation and the technical specifications of the basic components of the system are summarized in Table 1. The upper pressure level of the SORC was determined by considering the capability to use an open-drive scroll expander, which is commonly proposed for small scale applications such as the one investigated in the current study. The preference towards this type of expander instead of an expander of the hermetic type is justified by its lower cost. Moreover, high pressures (beyond 40 bar) inhibit the use of plate heat exchangers, which are relatively cheap and provide a very attractive heat transfer area-to-volume ratio. The maximum temperature limit of the working fluid of the SORC is imposed by the maximum temperature of the heating oil and the pinch point value of the heater that are assumed to be equal to 120 °C and 10 K respectively. Moreover, it is known that the thermal efficiency of ORCs increases as the condensation temperature decreases. The impact of the condensation temperature on the COP of the VCC is similar. However, it is not possible to decrease this temperature below a certain limit. On one hand, the condensation temperature cannot be lower than the temperature of the cooling water. On the other hand, since the system is intended to produce useful heat, the temperature of the water at the condenser outlet must be at least equal to 50 °C.

Table 1: Heating system and SORC assumptions

Biomass boiler-Heat Transfer Loop	
Boiler efficiency	82.9 %
Biomass Low Heating Value	16920 kJ/kg
Heat duty	85 kW_{th}
Heating oil	BP Transcal N
Heating oil maximum temperature	120 ºC
Heating oil temperature difference in boiler	15 K
SORC module	
Pinch point heater	10 K
Pinch point condenser	5 K
Maximum pressure of working fluid	40 bar
Maximum temperature of working fluid	110 °C
Isentropic efficiency of pump	50 %
Isentropic efficiency of expander	65 %
Electromechanical efficiency	85 %
Cooling water temperature at the condenser inlet	
Summer	30 °C
Winter	20 °C
Minimum cooling water temperature at the condenser outlet	40 °C
VCC module	
Pinch point evaporator	5 K
Nominal cooling load	4 kW_{th}
Evaporation temperature	10 °C
Condensation temperature	50 °C
Isentropic efficiency of compressor	75 %

4. WORKING FLUID SELECTION AND THERMAL EFFICIENCY MAXIMIZATION

The selection of the most appropriate working fluid is the first goal of the thermodynamic optimization. In the design process of the experimental unit, the screening method, which is commonly used in the literature, was followed. The two primary criteria used for selecting working fluid candidates are the critical temperature and pressure. Given the operational range of the temperatures and pressures of the system, as given in Table 1, a short list of 11 working fluids was created. These fluids are given in Table 2.

A second group of fluids were chosen from the working fluids of Table 2, by taking into account their Ozone Depletion Potential (ODP), their Global Warming Potential (GWP) as well as their ASHRAE safety group categorization. More specifically, R123 has a high ODP and is to be substituted by HFE7000 by 2030, R1234yf, R143a, R41, Propylene and n-Propane were rejected because of their high inflammability, R134a because of its high critical pressure and R161 because of its very limited market availability. Thus a final list of three working fluids, R125, R227ea and R404a, is assembled (in bold format in Table 2). The optimal working fluid is ultimately chosen by performing a thermodynamic investigation of their performance. The purpose of the optimization process is to determine which fluid exhibits the highest thermal SORC efficiency, when varying certain independent operation variables within the acceptable ranges specified in Table 1. The independent variables are the maximum pressure and temperature of the SORC and the condensation temperature. The evaporation temperature of the VCC is set at 7.5 °C, while a cooling load of 0 (cogeneration mode) and 3 kW (trigeneration mode) is assumed. The results of the optimization process for the three final working fluids are summarized in Table 3.

Table 2 : Preliminary selection of working fluid candidates (P_{crit}<40bar, T_{crit}<110^oC)

Working fluid	Critical temperature °C [11]	Critical pressure (bar) [11]	ODP [12]	GWP [12]	ASHRAE safety group [13]
R125	**66.02**	**36.18**	**0**	**3500**	**A1**
R134a	101.06	40.59	0	1430	A1
R143a	72.71	37.61	0	4470	A2L
R1234yf	94.70	33.82	0	4	A2L
R227ea	**101.75**	**29.25**	**0**	**3220**	**A1**
Propylene	92.42	46.65	0	1.8	A3
R41	44.13	58.97	0	92	---
N-Propane	96.70	42.48	0	3.3	A3
R161	102.22	47.02	0	12	--
R410a	72.80	48.60	0	2088	A1
R404a	**72.07**	**37.32**	**0**	**3300**	**A1**

As can be seen from Table 3, R227a exhibits both the highest SORC efficiency and the highest COP among the fluids examined. For this reason, it is selected for the experimental unit. It deserves to be noted that, despite it being commonly investigated as a working fluid for ORC applications in the literature [3, 14, 15], R227ea has not been very often used in actual ORC plants or experimental rigs. Moreover, it is rarely used in commercial refrigeration applications. Its market availability is relatively scarce and its cost significantly high, since its price is about eight times the price of R134a.

Table 3 Optimization results of the SORC thermal efficiency for the three final working fluids

	R125	R404a	R227ea
P_{max} (bar)	40	40	30.4
T_{max} (°C)	110	110	110
T_{cond} (°C)	50	50	50
T_{cool} (°C)	10	10	10
$\eta_{th,orc}$	2.16	2.81	4.92
$\eta_{el,net}$ (Q_c=0)	0.71	1.17	2.72
$\eta_{el,net}$ (Q_c=1)	0.23	0.73	2.30
COP	3.24	3.57	3.78

It can be thus considered as a "novel" working fluid. Based on the optimization results, the operational data of the system are summarized in Table 4.

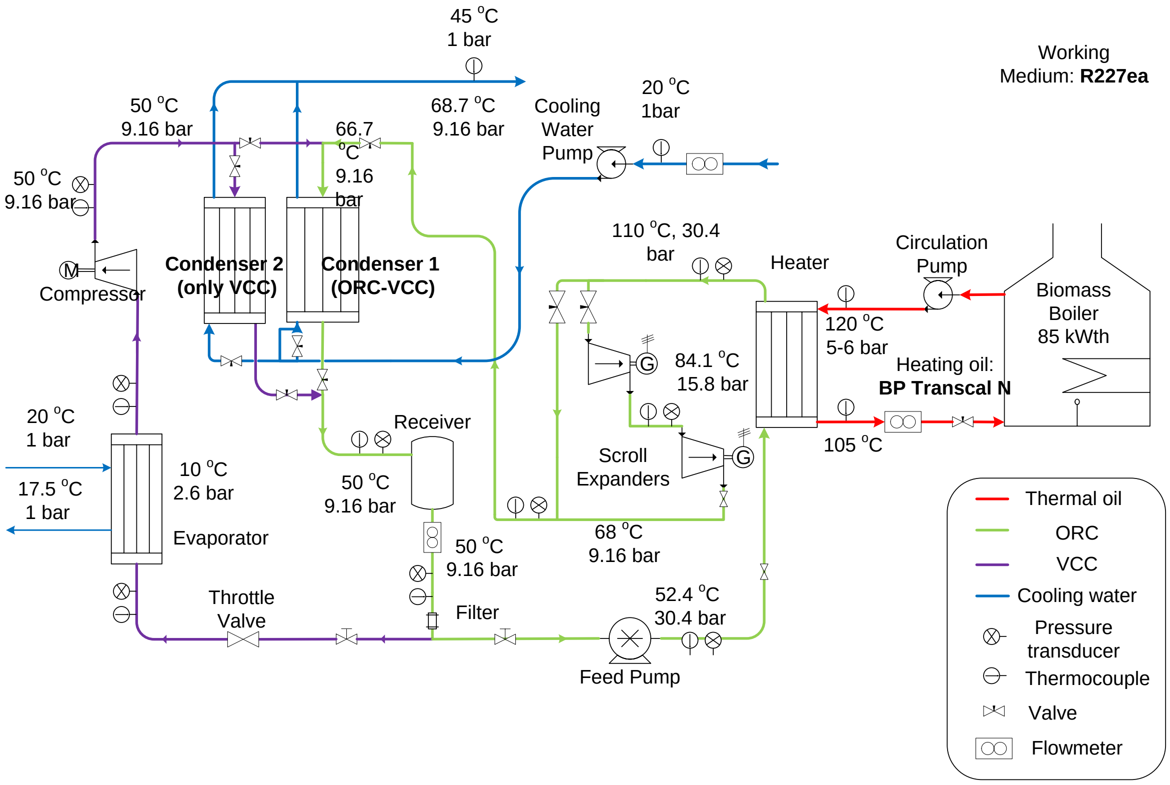

Figure 3: Process flow diagram of the experimental trigeneration facility

5. EQUIPMENT SELECTION

5.1 Overview

A process flow diagram of the experimental unit to be constructed along with operational data and design pressures and temperatures is presented in **Error! Reference source not found.**. The right part of the figure depicts the heating oil circuit along with the biomass boiler (red color) and the left part depicts the VCC (purple color). The SORC is in between these two circuits (green color). Two streams of water (blue color) are used for the cooling and the heating of the working fluid in the Condensers 1 and 2 as well as in the evaporator of the VCC respectively.

As can be seen, it was decided that two identical scroll expanders serially positioned will be used for the expansion process of the working fluid. Each one of these expanders is directly coupled to a generator. This is because of the limited volume flow ratio that is technically possible for these volumetric expanders. According to the literature [16], its value must not be higher than 5. Following a more conservative approach, the volume flow ratio for the expanders of the experimental rig is set between 2 and 3. During the start-up of the operation of the system, the working fluid does not flow through the expanders but goes through a by-pass loop until the vapor reaches the desired pressure and temperature.

Table 4: Design point operational data of the system at trigeneration mode

SORC module	
Mass flow rate	0.699 kg/s
Maximum pressure/temperature	30.4 bar/110 °C
Condensation pressure/temperature	9.16 bar/50 °C
Pressure/temperature after expanders	9.16 bar/68 °C
SORC net electric power output	2.40 kW_e
Overall net electric power output	0.99 kW_e
VCC module	
Mass flow rate	0.056 kg/s
Evaporation pressure/temperature	2.79 bar/10 °C
Condensation pressure/temperature	9.16 bar/50 °C

Pressure/temperature after compressor	9.16 bar/50 °C
Nominal cooling load	4 kW_{th}
Cooling system	
Water inlet temperature (summer/winter)	30 °C/20 °C
Water outlet temperature	45 °C
Mass flow rate	0.730 kg/s
Nominal heating load	85.8 kW_{th}
Heating system	
Heating oil inlet temperature	120 °C
Heating oil outlet temperature	105 °C
Mass flow rate	3.12 kg/s

Furthermore, the system includes two condensers, a large and a small one. The large one is intended for the trigeneration and cogeneration modes, during which the SORC is operational. Due to the significantly lower mass flow rate of the VCC, it is necessary to install a second small condenser, in the case that only the VCC is operational.
A description of the selection process of the equipment of the unit follows.

5.2 Boiler-heating oil circuit
The boiler has the capability to use both biomass pellets and natural gas as combustion fuel. The addition of gas burners is made in order to ensure the versatility of the system and its flexible response to rapidly varying loads. The gas burner consists of two stages. Its heat duty can be varied between 65 to 85 kW_{th} and its maximum fuel flow rate is 6.5 to 18.9 Nm^3/h. The flue gas temperature is between 220-230 °C. The pellet burner heat duty ranges between 40 to 80 kW_{th} and its fuel consumption rate between 8 to 20 kg/h. The wood pellets to be used as fuels must have a size around 6-8 mm.
The heating oil circuit includes a centrifugal pump of fixed rotational speed and volume flow rate and is able to handle fluids of temperatures of up to 350 °C. Moreover, the heating oil is stored in a round expansion vessel with a capacity of 250 liters.
The heating system will be automatically controlled through an electronic control panel. The user adjusts the desired heating oil maximum temperature. Since the volume flow rate of the heating oil has a fixed value, the control of this temperature is achieved by automatically shutting down starting up the burners of the boiler based on the readings of a thermostat. In order to investigate the performance of the system at off-design conditions, it is possible to change the heat duty of the burners.

5.3 Heat exchangers
The system includes four heat exchangers: a heater for the heating of the working fluid of the SORC by the heating oil, two cooling water condensers for condensing the working fluid (the requirement for two condensers was explained previously), as well as an evaporator for the VCC module, where the cooling output of the system is produced. All the heat exchangers to be installed are of the plate type. This is because these types of heat exchangers have a number of advantages compared to other types (i.e. shell and tube) for the operational range (pressures, temperatures, heat duty) of the unit under construction. Their advantages include the efficient heat transfer because of the development of high heat transfer coefficients between the fluid streams low volume to heat transfer area ratio and low cost. All heat exchanger models, except for the supercritical refrigerant-heating oil heater were readily proposed on-demand from manufacturers based on their technical specifications. However, most manufacturers were not able to dimension the supercritical heater of the working fluid. This is in part because of the lack of experience with the specific fluid (R227ea) and because of the supercritical conditions of the heat transfer. For this reason, a MATLAB code was used in order to estimate the required heat transfer surface of the heater. Geometrical data of commercially available heat exchanger models were input to the program and the heat transfer area was estimated using the heat exchanger partitioning method [17] and implementing method presented in [18] for calculating the overall heat transfer coefficient. Through a loop computational procedure, a specific plate heat exchanger model with defined number of plates was tested in order to estimate if its available heat

transfer surface would be sufficient for the process. If the available heat transfer surface was estimated to be lower than the required surface, more plates were added and a new loop started. The loop ended when the number of plates ensured that the heat transfer area of the heat exchanger would be higher than the required one. A safety factor was also considered, in order to ensure that the heat exchanger surface would suffice for completing the heat transfer under the desired conditions. The technical characteristics of all heat exchangers are summarized in Table 5.

Table 5: Technical specifications of heat exchangers

Heat exchanger	SORC heater		VCC evaporator		Condenser 1 (large)		Condenser 2 (small)	
Side	Hot	Cold	Hot	Cold	Hot	Cold	Hot	Cold
Fluid	BP Transcal N	R227ea	Water	R227ea	R227ea	Water	R227ea	Water
Mass flow rate (kg/s)	3.12	0.699	0.382	0.0522	0.643	0.737	0.067	0.062
Inlet pressure (bar)	5.5	30.4	1.013	2.58	9.115	1.013	9.126	1.013
Pressure drop (bar)	-	-	0.0073	0.007	0.062	0.026	0.0051	0.007
Inlet temperature (°C)	120	53	20	8.48 (x=0.4)	66.7	20	50	20
Outlet temperature (°C)	105	110	17.5	7.5	47.336	45	47.407	45
Heat duty (kW_{th})	85		4		77		6.5	
Heat transfer area (m^2)	4.71		0.39		4.32		0.31	

5.4 SORC expanders and VCC compressor

As stated previously, the expanders and the compressor are of the scroll type. Scroll expanders and other positive displacement type machines have been consistently proposed in the literature as ideal for ORCs with outputs in the scale between a few hundred Watts to 10 kW [19-22]. The expanders to be used in the SORC of the experimental unit are commercially available scroll compressors used in trucks, which are modified to enable their reverse operation as expansion machines. The most important considerations to be taken into account when selecting scroll expanders are their maximum operating pressure, their volume flow rate which greatly impacts their rotational speed, the latter being also directly influenced by their swept volume and their expansion ratio. The decision to use open drive scroll expanders instead of hermetic ones was made because of the lower cost and customization capability of the former versus the latter. Two same expanders were chosen to operate at different speeds. The reason for this choice was the fact that this commercial model of scroll compressor has a well-documented performance in expansion mode. The selection of the VCC compressor was simpler, since the operation data of the working fluid at its design point are typical for scroll compressors available in the market.

Table 6: Technical specifications of SORC expanders and VCC compressor

	Built-in swept volume (cc/rev)	Maximum rotational speed (rpm)	Rotational speed at design point (rpm)	Volume flow ratio at design point	Power at design point (kW)
SORC HP expander	121.1	10000	1384	2.47	2.86
SORC LP expander	121.1	10000	2632	1.90	2.89
VVC compressor	53.9	10000	1755	3.52	1.06

5.5 SORC pump

The feed pump increases the pressure level of the working fluid before its entrance to the heater. It is also used to control its mass flow rate. This control is achieved by varying the rotational speed of the motor coupled with the pump. The pumps that are commonly used and proposed in ORC units are of the positive displacement type [4]. For this type of pumps, the flow rate is almost proportional to their rotational speed. For the experimental unit, a diaphragm pump will be used. The SORC pump will have a design power output of 2.2 kW at 1450 rpm, which corresponds to a working fluid volume flow rate of around 29 lt/min. As can be seen in **Error! Reference source not found.**, the inlet and outlet pressures of the pump are 9.2 bar and 30.4 bar respectively. The pump is coupled with an electric motor of a nominal power of 3 kW.

5.6 Receiver

The receiver is a feed tank that is positioned after the condensers of the system. It provides a buffer storage of the working fluid during the operation of the unit, since only a certain percentage of its total quantity is circulating through the equipment components and the tubes at any given time (depending on the operating conditions). The feed tank also serves another purpose: it ensures that the working fluid at its outlet is at liquid state and has zero vapor fraction. For the sizing of the working fluid the total mass of the working fluid flowing through the equipment and piping was estimated. The tube diameters at each location of the unit were estimated by considering the proper fluid velocities (1-2 m/s for liquid and 8-10 m/s for gases). An additional volume for the sizing of heat exchangers, as well as an overall safety factor were taken into account. The total required volume of the receiver was thus estimated at around 50 lt.

5.7 System control equipment

The automated control of the system will be realized with the implementation of Programmable Logic Controllers (PLC). The PLC will be programmed to open and close the valves of the system. The PLC will also govern the coupling and the de-coupling between the generators and the scroll expanders. Moreover, they will be responsible for adjusting the rotational speed of the feed pump motor and the generators of the expanders, by the use of power inverters.

6. CONCLUSIONS

The thermodynamic optimization procedure (selection of working fluid, expander inlet pressure and temperature) of a trigeneration experimental unit combining an ORC with a VCC based on the heat input of a 85 kW_{th} biomass/natural gas boiler was developed. The working fluid found to be optimal for the unit is R227ea, while the optimal pressure and temperature are 30.4 bar and 110 ºC. At its design point (trigeneration mode), the system will produce 0.99 kW_e, a heating load of 85.8 kW_{th} and a cooling load of 4 kW_{th}.

The selection process of the basic equipment (biomass/natural gas boiler/heating oil circuit, heat exchangers, expanders/compressor, feed pump, receiver and system control equipment) of the experimental unit was also presented. The construction of the experimental system will be completed by the end of 2015 year and measurements on its operation will be carried out in order to evaluate its performance.

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NOMENCLATURE

η	efficiency	
P	power	kW
Q	heat duty	kW
P	pressure	bar
S	entropy	kJ/K
T	temperature	°C

Subscript

CHP	combined heat and power
comp	compressor
cond	condenser
cool	cooling
crit	critical
el	electric
heat	heat
in	input
max	maximum
net	net value
ORC	ORC working fluid
sys	system
th	thermal

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